INL/EXT-05-00228

Analysis Methods and Desired Outcomes of System Interface Heat Transfer Fluid Requirements and Characteristics Analyses

C. B. Davis

April 2005

Idaho National Laboratory

The INL is a U.S. Department of Energy National Laboratory
operated by Battelle Energy Alliance

INL/EXT-05-00228

Analysis Methods and Desired Outcomes of System Interface Heat Transfer Fluid Requirements and Characteristics Analyses

C. B. Davis

April 2005

Idaho National Laboratory

Idaho Falls, Idaho 83415

Prepared for the
U.S. Department of Energy
Assistant Secretary for Nuclear Energy
Under DOE Idaho Operations Office
Contract DE-AC07-05ID14517

ABSTRACT

The interface between the Next Generation Nuclear Plant (NGNP) and the hydrogen-generating process plant will contain an intermediate loop that will transport heat from the NGNP to the process plant. Seven possible configurations for the NGNP primary coolant system and the intermediate heat transport loop were identified. Both helium and liquid salts are being considered as the working fluid in the intermediate heat transport loop. A method was developed to perform thermal-hydraulic evaluations of the different configurations and coolants. The evaluations will determine which configurations and coolants are the most promising from a thermal-hydraulic point of view and which, if any, do not appear to be feasible at the current time. Results of the evaluations will be presented in a subsequent report.

CONTENTS

TABLES

FIGURES

1. INTRODUCTION

The interface between the Next Generation Nuclear Plant (NGNP) and the hydrogen-generating process plant will contain an intermediate loop that will transport heat from the NGNP to the process plant. The basic configuration of the intermediate heat transport loop has yet to be defined. For example, the loop may transport all or just a fraction of the nuclear plant's heat to the process plant. If the loop transports only a fraction of the heat to the process plant, the remainder of the heat will be used to produce electricity. Both direct and indirect electrical cycles are currently being considered. Consequently, a variety of connections between the primary coolant system and the intermediate heat transport loop are possible. Furthermore, both helium and liquid salts are being considered as the working fluid in the intermediate heat transport loop. The thermodynamic states of the working fluid have also not been selected.

Seven possible configurations for the NGNP primary coolant system and the intermediate heat transport loop have been identified. The ultimate objective of the program is to evaluate the advantages and disadvantages of each of the configurations and working fluids so that a specific design option can be recommended. However, the recommendation of a specific design requires input from a variety of disciplines related to materials, thermal-hydraulics, economics, safety, and plant operability. The purpose of this work is to evaluate the advantages and disadvantages of the configurations and working fluids from a thermal-hydraulic point of view to provide input to the decision making process. The work will also identify key research and development issues for the interface system.

This document describes the method that will be used to perform the thermal-hydraulic evaluations of the different configurations and coolants. The application of the method to generate results will be described in a subsequent report. The desired outcome of the analysis is to determine which of the configurations are currently feasible and to eliminate any that are not from further consideration. The relative advantages and disadvantages of the remaining configurations will be evaluated to determine which of the options appears to have the best chance for resulting in a successful design.

The method used in this analysis will: (1) identify key requirements of the NGNP and the hydrogen production plant that affect the intermediate heat transport loop; (2) describe and justify key assumptions used in the evaluation; (3) identify possible configurations of the intermediate heat transport loop; (4) perform preliminary stress evaluations to determine allowable pressures and required component thicknesses based on the creep rupture strengths of currently available materials; (5) estimate the size and thermal-hydraulic performance of various components in the intermediate heat transport loop including the heat exchangers and loop piping; and (6) compare and contrast the different options to aid in the selection of the configuration and working fluid.

The thermal-hydraulic evaluations of the different configurations and coolants that will be performed during the subsequent analysis are closely related to other work funded by the GenIV Energy Conversion Program. The evaluations described here will concentrate on determining component sizes, system flow rates, and thermodynamic states for the intermediate and process heat exchangers and heat transport loop piping. The Energy Conversion Program will determine the thermodynamic states for the remainder of the power conversion system and the overall cycle efficiencies for the same configurations described here. The cycle efficiencies will significantly affect the relative economics of the different design options. The results of the thermal-hydraulic and cycle efficiency evaluations are closely integrated. It is planned that results from both evaluations will be combined into the subsequent report.

The remainder of this report describes key requirements and assumptions used in the analysis, the seven configurations to be evaluated, and the methods used to perform the preliminary stress calculations and size the components.

2. KEY REQUIREMENTS AND ASSUMPTIONS

Two top-level temperature requirements have been identified for the interface system. These requirements are defined by the outlet temperature of the NGNP and the maximum temperature delivered to the hydrogen plant.

Previous evaluations have generally assumed that the outlet temperature of the NGNP is 1000 °C (MacDonald et al. 2003 and INEEL 2005). However, the Independent Technology Review Group (ITRG) (2004) recommended that the outlet temperature be reduced so that the maximum metal temperature was less than 900 °C. The primary basis for this recommendation was that "… there is high probability that heat exchanger technology will be available in time for deployment" with an outlet temperature of 900 °C, whereas the development of "… a licenseable nuclear system is not achievable within the NGNP schedule" for a 1000 °C outlet temperature and a 60-year design life. Furthermore, "… increasing the metal temperature beyond 900 °C will result in allowable stresses that make the use of metallic materials impractical." The ITRG recognized that higher outlet temperatures could be pursued once a successful demonstration of the NGNP had been achieved at a temperature limit of 900 °C.

For this analysis, the NGNP outlet temperature will be set at 900 °C, consistent with the recommendation of the ITRG (2004). Parametric calculations will be performed with higher outlet temperatures to determine their effects on component performance.

The efficiency of the hydrogen-production process increases with temperature. According to Argonne National Laboratory – West (ANLW) (2004), the temperature supplied to the hydrogen production plant should be at least 850 °C. The ITRG (2004) states that the current SI-based processes can operate with a temperature of at least 800 °C. For this analysis, the maximum temperature supplied to the hydrogen plant is assumed to be 850 °C

In order to provide estimates of component performance, assumptions are required about the basic configuration and operating conditions of the NGNP, the intermediate heat transport loop, and the hydrogen production plant. The primary assumptions are described below and summarized in Table 1. Parametric calculations will be performed to determine the effects of changes from the basic configuration.

The NGNP is assumed to produce 600 MW of thermal power. The NGNP is assumed to use helium coolant because the ITRG (2004) judged that it was not practical to develop and implement a reactor cooled by liquid salt by 2020. The nominal rise in fluid temperature across the core is assumed to be 400 °C, based on the point design (MacDonald et al. 2003). The nominal reactor pressure is assumed to be 7 MPa (INEEL 2005). The pressure drop across the hot stream of the intermediate heat exchanger (IHX) is assumed to be 0.05 MPa. This value is the same as the pressure drop across the core in the Gas Turbine-Modular Helium Reactor (GT-MHR) (General Atomics 1996). Since the pumping power associated with this pressure drop across the core was considered acceptable in the GT-MHR, the pumping power associated with this pressure drop across the IHX should also be acceptable.

The intermediate heat transport loop is assumed to receive 50 MW of thermal power (ANLW 2004). Parametric calculations will also be performed in which the total output of the reactor (600 MW) is assumed to be used for hydrogen production.

Estimates for the required separation distance between the nuclear and hydrogen plants depend on the design and safety criteria and vary considerably. For example, Verfondern and Nishihara (2004) calculated 300 m for the High-Temperature Engineering Test Reactor in Japan whereas Sochet et al. (2004) recommended 500 m for the High-Temperature Reactor. For this analysis, a nominal value of 300 m will be used, with parametric variations between 100 and 500 m to account for uncertainty. Subsequent to the start of this analysis, Smith et al. (2005) recommended a separation distance of from 60 to 120 m for the NGNP and the hydrogen production plant. The separation distance primarily affects the diameter of the hot and cold leg pipes in the heat transport loop. The final report will quantify the effect of the reduced separation distance on the diameters.

Table 1. Analysis assumptions.

Parameter	Nominal Value
NGNP:	
Power, MW	600
Outlet temperature, °C	900
Core temperature rise, °C	400
Pressure, MPa	7
IHX pressure drop, MPa	0.05
Intermediate heat transport loop:	
Power, MW	50
Separation distance, m	300
Hydrogen plant:	
Maximum delivered temperature, °C	850

The heat exchanger that thermally connects the NGNP to the heat transport loop is assumed to be a compact heat exchanger of the type designed by HEATRIC (Dewson and Thonon 1993). The heat exchanger that connects the heat transport loop to the hydrogen production plant is assumed to be a tube-in-shell heat exchanger with the heat transport fluid flowing on the shell side. This configuration allows the tubes to contain the catalysts necessary for hydrogen production, which is judged to be the most convenient configuration. The tube-side is assumed to be at low pressure (< 1 MPa) because the hydrogen production process is less efficient at high pressure. The hot and cold legs of the intermediate loop are assumed to be separate pipes, as opposed to an annular configuration. The purpose of these calculations is to compare the relative size of components between configurations. These calculations are not intended to achieve a final design for any configuration or to recommend one type of heat exchanger over another.

3. DESIGN CONFIGURATIONS

Up to seven plant configurations will be evaluated as part of this task. For convenience, the following definitions are used relative to the heat exchangers. The first heat exchanger downstream of the NGNP outlet is referred to as the IHX. The heat exchanger that connects the intermediate heat transport loop to the hydrogen production plant is referred to as the process heat exchanger (PHX). The third heat exchanger, which if present is located between the IHX and the PHX, is referred to as the secondary heat exchanger (SHX).

The seven plant configurations evaluated are illustrated in Figures 1 through 7. The configurations include direct and indirect electrical cycles with serial or parallel heat exchanger options. In the serial option, which is illustrated in Figures 1, 3, and 5, the IHX is located between the reactor outlet and the power conversion unit. In the serial option, the IHX removes less than 10% of the reactor power and directs it towards the hydrogen production plant. With this configuration, the hydrogen production plant receives a higher temperature fluid than the power conversion equipment. This configuration is relatively simple and is especially suitable for the demonstration of hydrogen production. However, the overall efficiency of the electrical production process is reduced. In the parallel heat exchanger option, which is illustrated in Figures 2, 4, 6, and 7, the hottest fluid is divided, with most going towards the power conversion unit and the remainder going towards the hydrogen production plant. This configuration is more complicated, but results in a higher overall efficiency because both the electrical and hydrogen production plants see the maximum possible temperature. With these options, a small compressor or blower is required to compensate for the pressure loss across the IHX and allow the fluid streams to mix downstream of the recuperator. The final option includes a SHX as shown in Figures 3, 4, 5, and 6. This option provides for additional separation of the nuclear and hydrogen plants, which should increase the safety of both plants and may make the nuclear plant easier to license. However, this option requires more capital investment and lowers the overall efficiency of the plant.

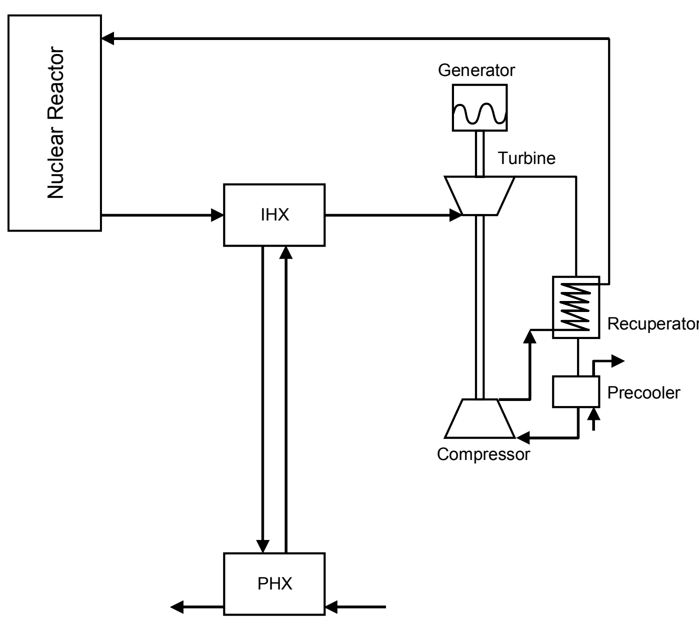

Figure 1. Configuration 1 (direct electrical cycle and serial IHX).

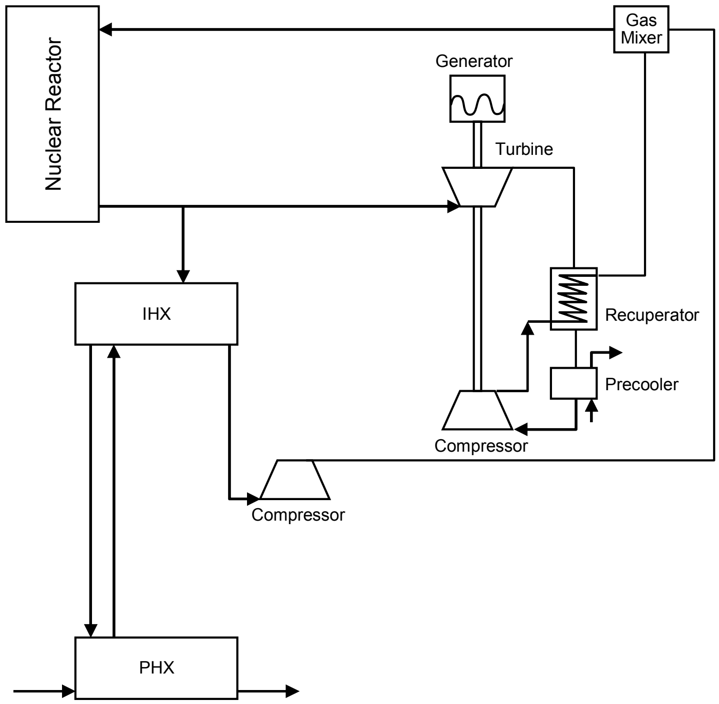

Figure 2. Configuration 2 (direct electrical cycle and a parallel IHX).

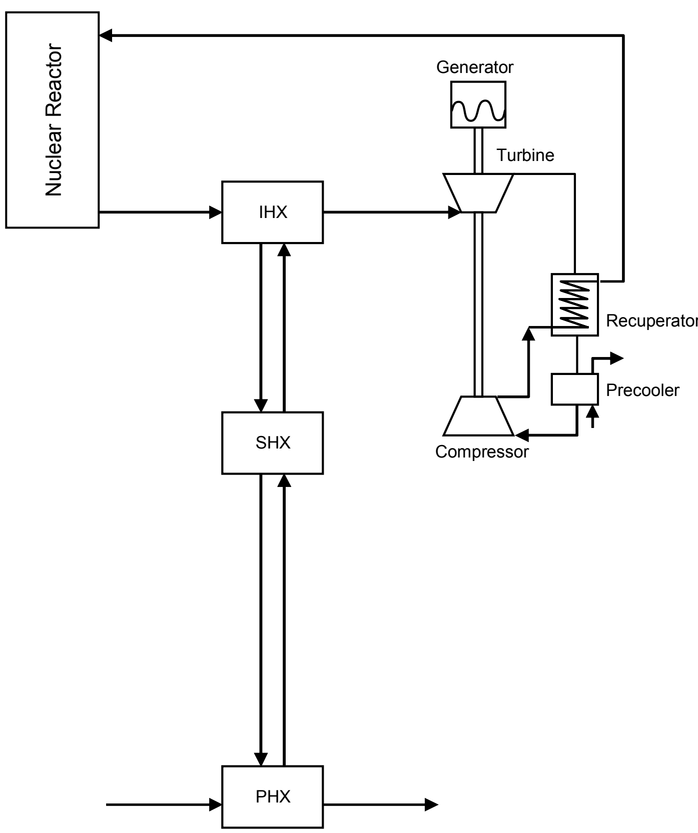

Figure 3. Configuration 3 (direct electrical cycle, serial IHX, and SHX).

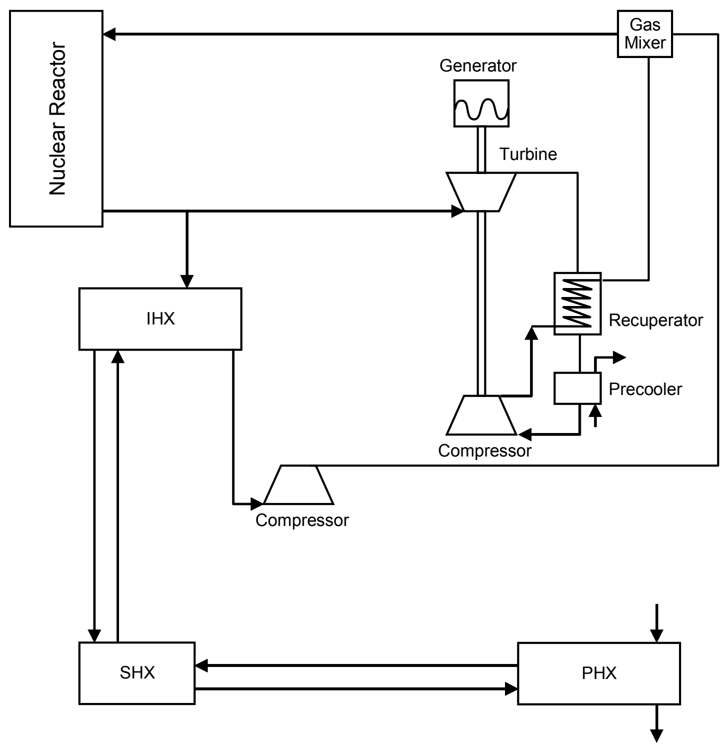

Figure 4. Configuration 4 (direct electrical cycle, parallel IHX, and SHX).

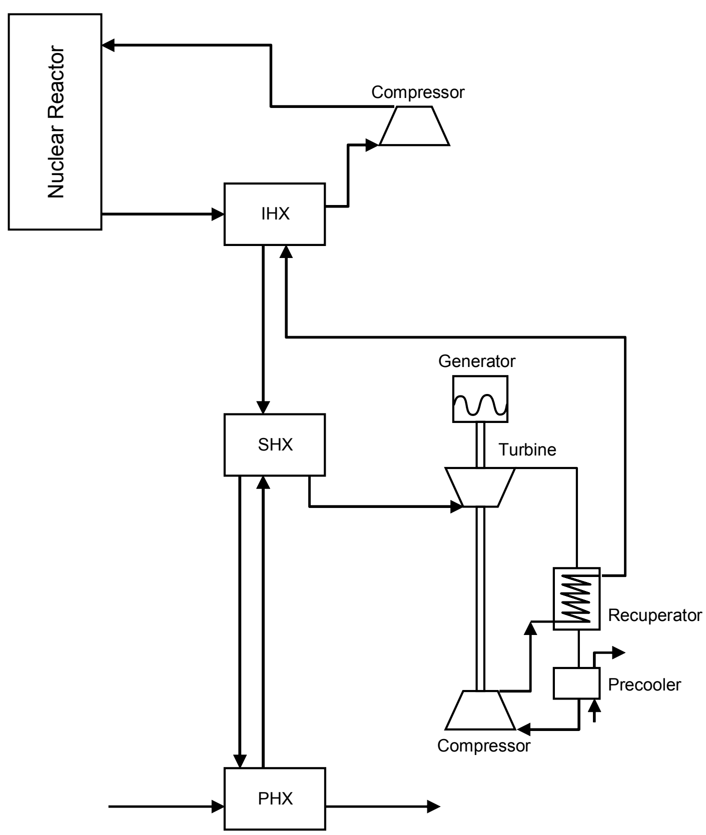

Figure 5. Configuration 5 (indirect electrical cycle and a serial SHX).

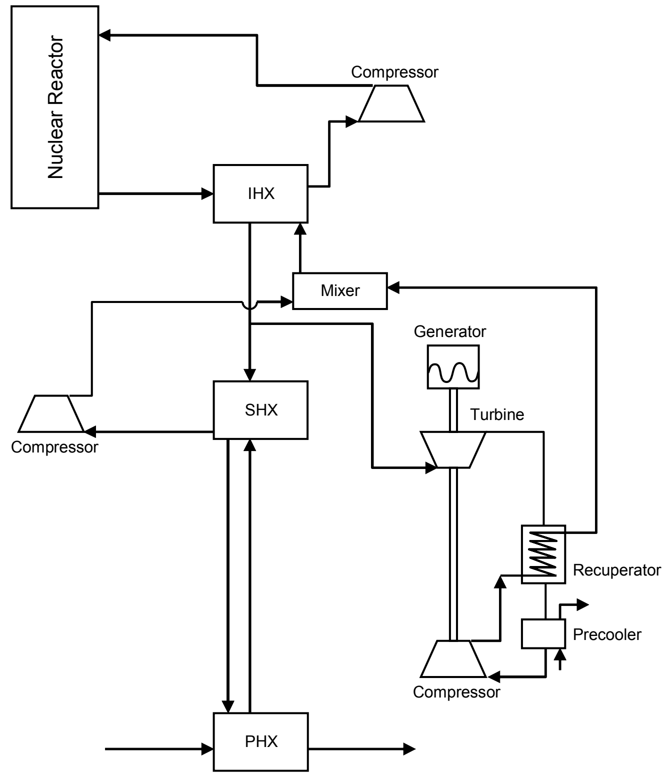

Figure 6. Configuration 6 (indirect electrical cycle and a parallel SHX).

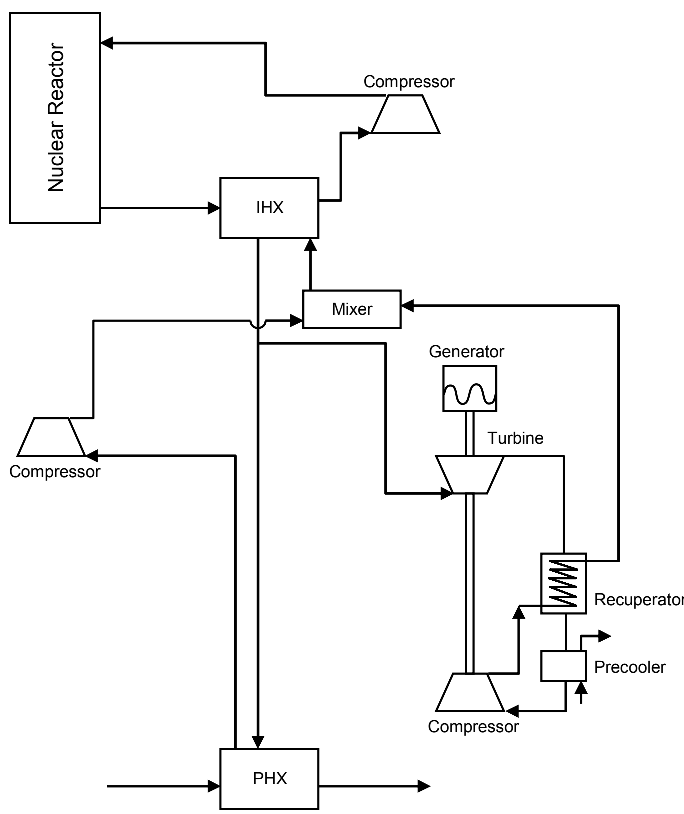

Figure 7. Configuration 7 (indirect electrical cycle and a parallel PHX).

4. STRESS ANALYSIS

A simplified stress analysis will be performed for different components in the various configurations. The analysis will determine the thickness required so that the circumferential stress is less than or equal to an assumed allowable value. The use of consistent stresses will allow identification of limiting components and a fair comparison between different configurations.

The creep rupture strength of a material depends on the operating time at a given temperature. Figure 8 shows that the rupture strength of Alloy 800 decreases sharply with temperature. At an operating time of 10^5 h (about 11 years), the rupture strength is 240 MPa at 500 °C, but decreases to 8 MPa at 900 °C. The rupture strength also depends on the time at temperature. At 900 °C, the rupture strength increases from 8 to 16 MPa when the operating time decreases from 10^5 to 10^4 h. The data presented in Figure 8 suggest that the mechanical design of the heat transport loop will be a challenge because of the desired high temperature and the long lifetime, both of which act to reduce the rupture strength.

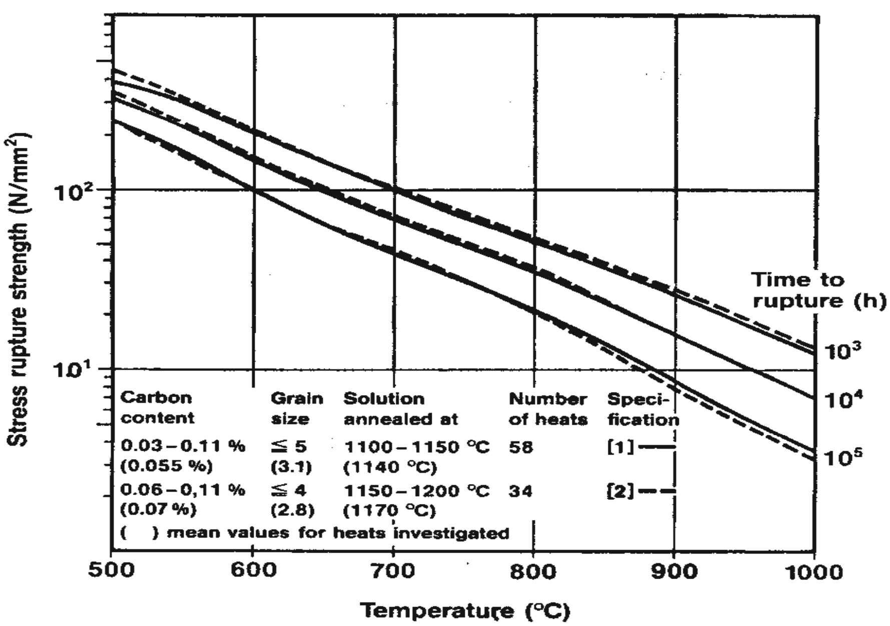

Figure 8. Creep rupture strength of Alloy 800 (from Diehl and Bodman 1990).

The creep rupture strengths of three candidate materials for the heat transport loop are shown in Figure 9 for a temperature of 900 °C. These three materials are Alloy 800HT, which is a high-temperature variation of Alloy 800 (Special Metals 2004a), Alloy 617 (Special Metals 2004b), and Hastelloy X (Haynes International 2005). Alloy 617 has the highest rupture strength of these three materials at 900 °C. The allowable stress will eventually be specified by an applicable code, but will be less than the strengths shown in Figure 9 to account for safety factors. For this analysis, the allowable stress will be assumed to be half of the creep rupture strength.

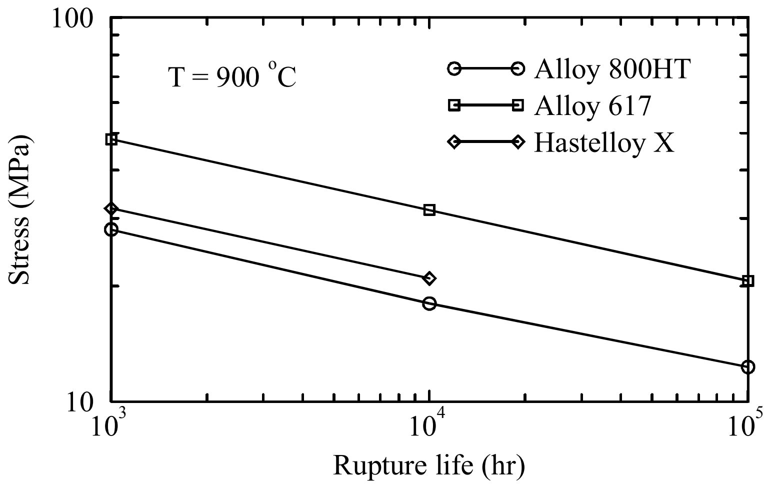

Figure 9. Creep rupture strengths of candidate materials.

A simple stress analysis will be performed to determine the required thickness for the heat transport loop piping and the heat exchangers. For thick walled cylinders, the tangential stress, σ, is calculated as (Crandall et al. 1972)

$$\sigma = \frac{P_i[(r_o/r)^2 + 1] - P_o[(r_o/r_i)^2 + (r_o/r)^2]}{(r_o/r_i)^2 - 1} \quad , \tag{1}$$

where r is the radius, P is the pressure, and the subscripts i and o refer to the inner and outer surfaces, respectively. The stress is negative if the external pressure exceeds the internal pressure, but the maximum magnitude always occurs at the inner surface. The radius ratio that causes the maximum stress to be less than or equal to the allowable stress, σ_D, can be calculated from Equation (1). For cases where the internal pressure exceeds the external pressure, the limiting ratio is

$$\frac{r_o}{r_i} \geq \sqrt{\frac{\sigma_D + P_i}{\sigma_D + 2P_o - P_i}} \quad . \tag{2}$$

For cases where the external pressure exceeds the internal pressure, the maximum, absolute value of the stress will be less than or equal to the allowable stress when the radius ratio is

$$\frac{r_o}{r_i} \geq \sqrt{\frac{\sigma_D - P_i}{\sigma_D - 2P_o + P_i}} \quad . \tag{3}$$

The maximum circumferential stress for the piping of the intermediate heat transport loop and the PHX will be calculated from Equation (1) for various internal and external pressures. The pressures will correspond to both low-pressure and high-pressure applications, and represent the use of either low-pressure helium or a liquid salt as the working fluid or high-pressure helium.

A simple stress analysis will also be performed for the IHX assuming that it is a compact heat exchanger of the type designed by HEATRIC (Dewson and Thonon 1993). The design of the heat exchanger channels is defined by the channel diameter, d, pitch, p, and plate thickness, t_p, as illustrated in Figure 10. Each plate contains either hot or cold fluid, but not both. Adjacent plates contain the other fluid. Following the method used by Dostal et al. (2004), the minimum wall thickness between channels, t_f, can be approximated as

$$t_f \geq \frac{p}{\frac{\sigma_D}{\Delta P} + 1} , \tag{4}$$

where σ_D is the allowable stress and ΔP is the differential pressure between the hot and cold streams. Expressing Equation (4) in terms of pitch-to-diameter ratio yields

$$\frac{p}{d} \geq 1 + \frac{\Delta P}{\sigma_D} . \tag{5}$$

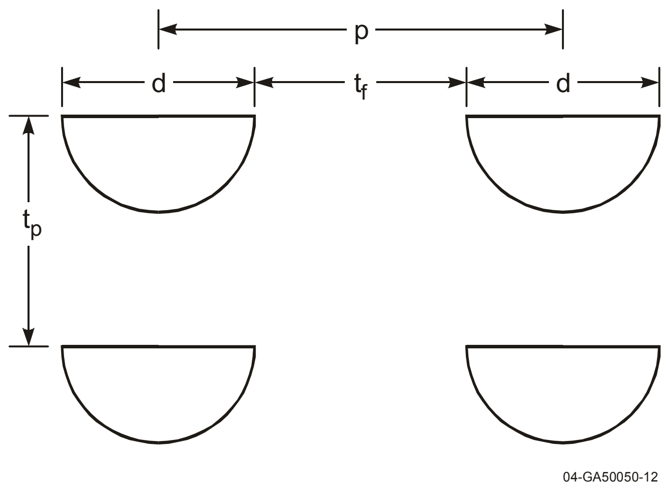

Figure 10. Illustration of IHX channels.

The required plate thickness can also be calculated based on the method of Dostal et al. (2004). The plate is assumed to be a thick-walled cylinder, with an inner radius of d/2 and an outer radius of t_p. Equations (2) and (5) are be used to calculate the thickness-to-diameter and pitch-to-diameter ratios for the IHX as a function of allowable stress and various pressures of the hot and cold streams. The allowable stress is assumed to be 10 MPa, which is approximately half of the rupture strength of Alloy 617 at 900 °C.

5. COMPONENT SIZES

The nominal temperature drop between the outlet of the NGNP and the maximum temperature delivered to the hydrogen production plant is 50 °C. This temperature drop imposes requirements on the effectiveness of the heat exchangers that connect the NGNP and production plant and the amount of heat loss than can be tolerated in the intermediate loop. In order to perform preliminary calculations, heat loss will be assumed to cause the temperature to drop 10 °C in the hot leg of the intermediate loop. Assuming the same geometry in the hot and cold legs of the intermediate loop, more heat is lost from the hot leg than from the cold leg. Based on nominal temperatures, the heat loss from the hot legs is expected to be about 70% of the total. The total temperature drop in the loop piping was assumed to be 10/0.7 = 14.3 °C, with 10 °C occurring in the hot leg and the remaining 4.3 °C occurring in the cold leg.

Although the total temperature drop between the NGNP and the production plant is fixed by assumption, the distribution of the temperature drop between the heat exchangers and heat loss can be varied. For example, if the heat loss can be reduced, smaller heat exchangers can be used. After accounting for heat loss, the remaining temperature drop between the outlet of the NGNP and the maximum temperature delivered to the hydrogen production plant is divided evenly between the IHX, PHX, and, if present, the SHX.

As mentioned previously, the temperature drop between the NGNP and the production plant imposes requirements on the heat exchangers. The effectiveness of a heat exchanger, ε , (Kreith 1964) can be calculated as

$$\varepsilon = \frac{(\dot{m}c_p)_h (T_{h\,in} - T_{h\,out})}{(\dot{m}c_p)_{min} (T_{h\,in} - T_{c\,in})} \tag{6}$$

where $\dot{m}$ is the mass flow rate, c_p is the specific heat capacity at constant pressure and is assumed constant, and T is the temperature. The subscripts h and c refer to the hot and cold sides of the heat exchanger, the subscripts in and out refer to the inlet and outlet ends of the heat exchanger, and the subscript min refers to the minimum value for the hot and cold sides.

The heat exchangers are assumed to be in counterflow, which requires less surface area than is required for parallel flow (Kreith 1964). Counterflow heat exchangers are therefore smaller, and presumably cheaper, than corresponding heat exchangers in parallel flow. If the values of $\dot{m}c_p$ are the same for the hot and cold streams, the effectiveness depends only on the inlet and outlet temperatures.

Estimates are also made to size the heat exchangers. The required heat transfer area, A_{ht}, can be calculated from equations given by Krieth (1964)

$$A_{ht} = \frac{\varepsilon(\dot{m}c_p)_{min} (T_{h\,in} - T_{c\,in})}{U\Delta T} \quad , \tag{7}$$

where U is the overall heat transfer coefficient and ΔT is the log-mean temperature difference, which is calculated as

$$\Delta T = \frac{\Delta T_a - \Delta T_b}{\ln(\Delta T_a / \Delta T_b)} \quad , \tag{8}$$

where ΔT_a is the temperature difference between the hot and cold fluid streams at one end of the heat exchanger and ΔT_b is the temperature difference at the other end. The overall heat transfer coefficient is calculated from the heat transfer coefficients on both sides of the exchanger and the thermal conductivity and thickness of the metal. The heat transfer coefficients and the thermal conductivity are assumed constant over the length of the heat exchanger. The heat transfer coefficients are calculated using the Dittus-Boelter correlation, with a leading coefficient of 0.021 for gases and 0.023 for liquids (INEL 1995). The thermal conductivity of the metal is calculated assuming Alloy 800, and varies between 18 and 26 W/m-K over the temperature range of interest.

The pressure drop across a component is calculated from either the Blausius equation (Bird et al. 1960) or the more accurate Zigrang-Sylvester correlation (INEL 1995) for turbulent flow and the exact solution for fully developed laminar flow in a tube (Bird et al. 1960).

The front face of the IHX is assumed to be square. Iterations are performed to determine the width of the IHX. First, a diameter of the semicircular flow channel is assumed. The pitch between channels and the plate thickness are then calculated from the ratios given by Equations (2) and (5). A width of the IHX is then assumed. The flow areas of the hot and cold streams are then calculated from the width and geometries of the channels and plates. The mass flow rates for both streams are calculated from an energy balance and the assumed inlet and outlet temperatures. The overall heat transfer coefficient and effectiveness are then calculated. The required heat transfer area is then calculated from Equation (7). The length of the heat exchanger is then calculated from heat transfer area and the wetted perimeter of the channels, which allows the calculation of the pressure drop. The heat exchanger width is then varied until the desired pressure drop is obtained.

A similar method is used for the tube-in-shell PHX. First, the tube inner diameter is assumed. The tube thickness is then calculated from ratios determined in the stress analysis. The pitch-to-outer-diameter ratio of the tubes is set to 1.3, a typical value for tube bundles. The tube bundle is assumed to have a triangular pitch. Details on the heat transfer coefficients and fluid temperature distribution on the process side of the PHX are not yet available. Consequently, the heat transfer coefficient on the process side is assumed to be 2000 W/m^2-K. The inlet and outlet fluid temperatures on the process side are also assumed. The inner diameter of the shell is then varied until the desired pressure drop was obtained.

The inner diameters of the hot and cold leg pipes in the heat transport loops are sized to produce a given pressure drop. The thickness of the piping is based on the results of the stress analysis. The heat loss is calculated using an overall heat transfer coefficient, which accounts for the thermal resistance of the heat transfer coefficient at the inner and outer surfaces, the pipe metal, and the insulation (Bird et al. 1960). Specifically,

$$U_0 = \frac{1}{r_0\left(\frac{1}{r_0 h_0} + \frac{\ln(r_1/r_0)}{k_1} + \frac{\ln(r_2/r_1)}{k_2} + \frac{1}{r_2 h_2}\right)} \tag{9}$$

where U_0 is the overall heat transfer coefficient based on the inner surface area of the pipe, k_1 and k_2 are the thermal conductivities of the pipe metal and insulation, respectively, and r_0, r_1, and r_2 are the radii of the inner surface of the metal, the outer surface of the metal, and the other surface of the insulation, respectively. The heat transfer coefficient at the inner surface, h_0, is calculated using the Dittus-Boelter correlation (INEL 1995) as described previously. The heat transfer coefficient at the outer surface, h_2, accounts for natural convection and radiation. The convective contribution is calculated using the Churchill-Chu correlation for natural convection from a horizontal cylinder (Holman 1986). The radiation term is calculated assuming that the pipe is in a large enclosure (Homan 1986), such as in a buried conduit. The thermal conductivity of the metal is based on Alloy 800. The thermal conductivity of the insulation is assumed to be 0.1 W/m-K, which is a representative value for glasswool. The thickness of the insulation is varied to obtain the desired heat loss. In case an alternate insulation material is eventually selected, the required thickness can be approximated by the thickness value reported here multiplied by the ratio of the actual thermal conductivity to the assumed thermal conductivity.

Estimates of the pumping power, Q_p, are approximated using

$$Q_p = \frac{\dot{m}\Delta P}{\rho} \quad , \tag{10}$$

where $\dot{m}$ is the mass flow rate, ΔP is the pressure drop, and ρ is the fluid density (Glasstone and Sesonske 1967). The fluid density is based on the temperature at the inlet to the reactor for the hot stream of the IHX and based on the temperature of the cold stream entering the IHX for the intermediate heat transport loop.

6. EVALUATION OF CONFIGURATIONS

The methods described in Sections 4 and 5 will be used to evaluate the configurations illustrated in Figures 1 through 7. The different configurations will be evaluated relative to their ability to meet expected allowable stresses based on existing materials, component size and performance, and cycle efficiency. The advantages and disadvantages of the different configurations and coolants will be identified. Parametric calculations will be performed to investigate the sensitivity of the calculated results to important parameters and assumptions.

The desired outcomes from these evaluations will be to determine which configurations and coolant options are considered the most promising and which appear not feasible at this time.

7. REFERENCES

ANLW, 2004, “Reactor/Process Interface Requirements,” ANL W7500-0001-ES-00, Revision 0, July.

Bird, R. B., W. E. Stewart, and E. N. Lightfoot, 1960, “Transport Phenomena,” John Wiley & Sons, Inc., New York.

Crandall, S. H, N. C. Dahl, and T. J. Lardner, 1972, “An Introduction to the Mechanics of Solids,” Second Edition, McGraw-Hill Book Company, New York.

Dewson, S. J and B. Thonon, 2003, “The Development of High Efficiency Heat Exchangers for Helium Gas Cooled Reactors,” Paper 3213, ICAPP03.

Diehl, H. and E. Bodman, 1990, “Alloy 800 Specifications in Compliance with Component Requirements,” Journal of Nuclear Materials, 171, pp. 63-70, 1990.

Dostal, V., M. J. Driscoll, and P. Hejzlar, 2004, “A Supercritical Carbon Dioxide Cycle for Next Generation Nuclear Reactors,” MIT-ANP-TR-100, March.

Krieth, F., 1964, “Principles of Heat Transfer,” International Textbook Company, Scranton.

General Atomics, 1996, “Gas Turbine-Modular Helium Reactor (GT-MHR) Conceptual Design Description Report,” GA Project No. 7658, 910720 Revision 1, July.

Glasstone, S. and A. Sesonske, 1967, “Nuclear Reactor Engineering,” Van Nostrand Reinhold Co., New York.

Haynes International, 2005, http://www.haynesintl.comXsite/Xcsrs.html.

Holman, J. P., 1986, “Heat Transfer,” Sixth Edition, McGraw-Hill Book Company, New York.

Independent Technology Review Group, 2004, “Design Features and Technology Uncertainties for the Next Generation Nuclear Plant,” INEEL/EXT-04-01816, June.

INEEL, 2005, “Next Generation Nuclear Plant Research and Development Program Plan,” INEEL/EXT-05-02581, January.

INEL, 1995, “RELAP5/MOD3 Code Manual,” Vol. 1 and 4, NUREG/CR-5535, INEL-95/0174.

MacDonald, P. E., J. W. Sterbentz, R. L. Sant, P. D. Bayless, R. R. Schultz, H. D. Gougar, R. L. Moore, A. M. Ougouag, and W. K. Terry, 2003, “NGNP Preliminary Point Design – Results of the Initial Neutronics and Thermal-Hydraulic Assessments,” INEEL/EXT-03-00870, July.

Sochet, I., J. L. Rouyer, and P. Hemmerich, 2004, “Safe Hydrogen Generation by Nuclear HTR,” Paper 4261, Proceedings of ICAPP ‘04, Pittsburgh, PA, USA, June 13-17.

Special Metals, 2004a, http://www.specialmetals.com/products/incoloyalloy800h&ht.htm, September.

Special Metals, 2004b, http://www.specialmetals.com/products/incoloyalloy617.htm, September.

Smith, C., S. Beck, and B. Galyean, 2005, “An Engineering Analysis for Separation Requirements of a Hydrogen Production Plant and High-Temperature Nuclear Reactor,” INL/EXT-05-00137 Rev 0, March.

Vernondern, K. and T. Nishihara, 2004, “Valuation of the Safety Concept of the Combined Nuclear/Chemical Complex for Hydrogen Production with HTTR,” JUEL-4135.